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EXPERIMENTAL AND THEORETICAL STUDY OF HEAT PUMP PERFORMANCE ENHANCEMENT BY USING A NANOREFRIGERANTS

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ABSTRACT

The effect of Al_2O_3 nanoparticles on the performance of heat pump to improve its operational efficiency was presented in this theoretical and experimental study. In the experimental work the heat pump charged with R600a inclusive with 0.06 % vol. of Al_2O_3 and used as a nanorefrigerant. Three different nanoparticles size 20nm, 40 nm and 50 nm of Al_2O_3 have been used for the preparation of nanolubricant in the present study. The theoretical approach includes simulations modeling the heat pump components such as compressor, evaporator, condenser and an expansion valve by computer of the heat pump system by using commercial MATLAB. The results showed that the addition of nanoparticles to the refrigerant will improve its characteristics of refrigeration system heat transfer and thermal properties. Also, it showed that the using nanorefrigerant in refrigeration system will work normally at all conditions employed in this work. The experimental results found that the heat pump coefficient of performance increased by 19.1%, but the power consumption reduced by 21.8 % when using a mineral oil with 20 nm nanoparticles size of Al_2O_3 instead of the conventional mineral oil only. Finally, the refrigeration effect increased and work of compressor decreased by using a small nanoparticles size of Al_2O_3.

Key words: Nano-Refrigerant, Heat pump, Coefficient of Performance, R600a.

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1. INTRODUCTION

Recently, the heat pumps used in many applications such as cooling and heating of air conditioning, hot water production, centers of computers, restaurants, public buildings, hotels, and so on. The vapour compression refrigeration systems were conducted some investigators to examine the effect of nanoparticle was added in refrigerant/lubricant. Most of the investigators' conclusions obtained a remarkable improvement in thermophysical of the refrigerant. When the working fluid in refrigeration cycle passes through the compressor, it will mix with lubricant whose contains Nano-sized particles dissolved with him and the working fluid after leaving the compressor is called Nanorefrigerant. Using the Nanorefrigerant in the refrigeration cycle has given some advantages such as improvement of the heat transfer coefficient, reducing the power consumption in the compressor, and enhancement of the refrigeration effect in the evaporator. Many works have been conducted in the past few decades. The experimental investigation for the effect of using working fluid represented by TiO_2-R600a nano-refrigeranton on the performance of a domestic refrigerator found that the refrigerator system operated efficiently and normally and the saving of energy was about 9.6% [1] while the study for the effect of using Nano Refrigerant of Al_2O_3-R134a in refrigeration system showed that the addition of Al_2O_3 nanoparticles leads to COP enhancement of the system by about 10.32% less energy with the concentration used [2]. The result of studying the effect of using CuO nanoparticles in R134a refrigerant in the system showed that the coefficient of evaporating heat transfer increases with the increase of CuO nanoparticles concentration ranged from 0.1% to 0.55 % and nanoparticles size ranged from 15 to 25 NM [3]. In domestic refrigerator, the results of enhancement of heat transfer by using the working fluid of R600a R600a/mineral with mineral oil and Al_2O_3 nanoparticles, the results showed a reduction in the power consumption and the freezing capacity was higher by 11.5 % when using a mixture of nanorefrigerant instead of Polyester (POE) [4]. The pure R134a and R134a with Al_2O_3 nanoparticles with 20 nm size with three mass of mass fraction of 0.04%, 0.06% and 0.08% wt. used in vapour compression refrigeration system. The results showed that the refrigeration system with nanorefrigerant works safely and normally, also, the power consumption reduces by 14.71% and freezing capacity is higher while the COP increases by 28.93% at 0.06% [5]. To reduce energy consumption in buildings, the modeling of air conditioning of direct expansion and heat pump systems can be used in developing energy saving methods. [6] Discussed the development and validation of an artificial neural network modeling technique to predict the performance of air conditioning and heat pump of type of a direct expansion. In this paper, the system performances of the heat pump by using (R600a) refrigerant with Al_2O_3 nanoparticle were considered experimentally and theoretically.

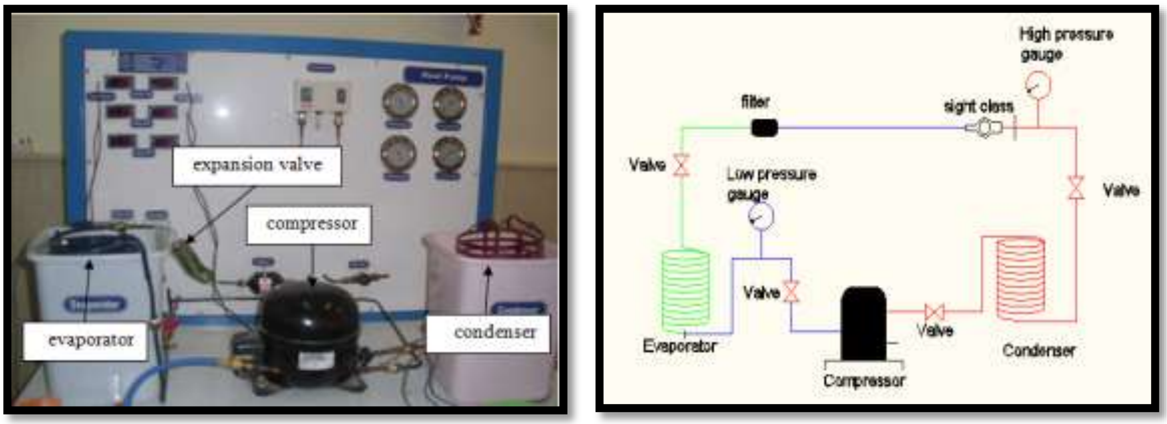
2. EXPERIMENTAL APPARATUS

Fig. (1) shows the main component in the test rig which consists of the compressor (Hermetic compressor), water cooled condenser made of copper, reversing valve, and an expansion valve (capillary tube), the cylindrical spiral coil evaporator was made of copper was completely immersed in water. The pressure at discharge and suction of compressor, outlet of condenser and at outlet of evaporator was measured by pressure gauges. The different point's temperatures of the experimental setup were measured using T-type thermocouples. For accurate measurements, six thermocouples were used to measure the temperature. The calibration of the thermocouples was done using a mercury thermometer. The power consumed by the system was measured using a digital Wattmeter. The setup of experimental was placed on a flat stage. The room temperature varied with temperature of ambient by ±1-3°C and the air flow speed was less than 0.3 m/s. Before charging the test rig with the

refrigerant, the system was checked properly for leaks by charging the system with N_2 gas at a pressure of 250 Psi, the system was evacuated by a vacuum pump to -30 Psi. The system was charged with the refrigerant (R600a) but the compressor was filled with nanolubricant. Four cases have been considered for this experimental study as illustrated in Table (1) knowing that the concentration of Al_2O_3 in the Nano-Refrigerant in each case of (2, 3and 4) were equal to 0.06%.

Table 1 Cases symbol notification.

Symbol	Cases
S_1	SUNISO 3GS (mineral oil) filled hermetic compressor
S_2	Mineral oil + 20nm nanoparticles size of Al_2O_3
S_3	Mineral oil + 40nm nanoparticles size of Al_2O_3
S_4	Mineral oil + 50nm nanoparticles size of Al_2O_3



a) b)

Figure 1 Photo and schematic diagram for the experimental rig**.**

2.1. Preparation of Nanolubricant

The particles of Al_2O_3 are commercially available spherical in shape. Three different nanoparticles size 20nm, 40nm and 50nm of Al_2O_3 have been used for the preparation of Nanolubricant in the present study. The nanorefrigerant with 0.06% concentration of Al_2O_3 and the refrigerant (R600a) were tested in the setup. The nanoparticles of Al_2O_3 and compressor oil mixture were prepared with aid of an ultrasonic vibrator [100 kHz, 300 W, Toshiba, England]. The mixture was remaining vibrated with an ultrasonic homogenizer for a period of not less than one hour to prevent any coagulation or deposition of particles in the mixture to obtain proper mixture homogenously. The experimental notice showed that the stable dispersion of alumina nanoparticles can be kept for more than 3 days without any deposition as shown in Fig.(2).

(a) Pure compressor oil lubricant

(b) Compressor oil nanolubricant.

Figure 2 Photographs of Nanolubricant**.**

2.2. Data Analysis

For the rate of heat transfer calculation from the condenser and the evaporator, the values of enthalpies at various points were plotted on the P-h diagram for refrigerant (R600a) as shown in **Fig.(3)**, which depends upon combination of the temperature and pressure, has been used to determine enthalpy values with Cool Pack software. The important factors which affect the performance of heat pump cycle are the Coefficient of Performance (COP), Heat–Rejection Ratio (H.R.R), and Energy consumption by the compressor [9]:

a) Coefficient of Performance(COP)

$$COP = \frac{Heat\ removal}{Work}$$

$$COP_{exp} = \frac{h_1 - h_4}{h_2 - h_1} \qquad (1)$$

b) Heat–Rejection Ratio (H.R.R)

$$HRR_{exp} = \frac{rate\ of\ heat\ rejection\ at\ the\ condenser}{rate\ of\ heat\ absorbion\ at\ the\ evaporator}$$

$$HRR_{exp} = \frac{h_2 - h_3}{h_1 - h_4} \qquad (2)$$

c) Energy consumption by the compressor.

$$W_{exp} = \dot{m} * (h_2 - h_1) \qquad (3)$$

3. HEAT PUMP MODELING

In this study, the approach considered includes simulations modeling by computer of the heat pump system by using commercial MATLAB. The modeling includes the conventional heat pump as shown in Fig.3 components such as compressor, evaporator, condenser and an expansion valve [10].

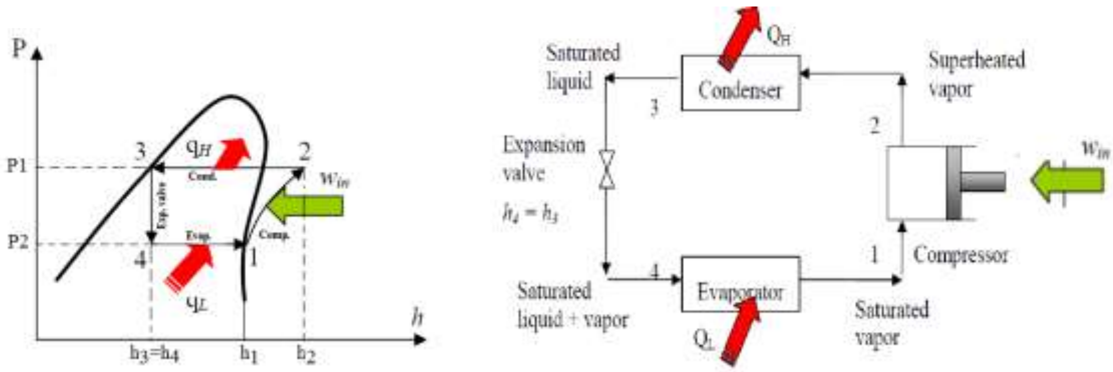


Figure 3 Pressure-enthalpy diagram of heat pump cycle.

A. Compressor model

In this procedure the following assumption made such as neglect the valves dynamic, there is no gas leakage. Thus, based on flow through orifice equations, the mass flow rates of the inlet and outlet given by [10]:

$$\dot{m}_{in} = \rho_{suc} C_{sv} S_{sv} \sqrt{\frac{2(P_{suc} - P_{cyl})}{\rho_{suc}}} \quad (4)$$

$$\dot{m}_{out} = \rho_{cyl} C_{dv} S_{dv} \sqrt{\frac{2(P_{cyl} - P_{dis})}{\rho_{cyl}}} \quad (5)$$

The pressures of the evaporator and condenser of the suction and discharge in addition of suction density along orifice coefficients and valves flow areas have to be known. Thus, the the compressor real mass per cycle can be defined as:

$$m = \eta_k C_{suc} V_{str} \quad (6)$$

Also, the compressor volumetric efficiency defined by:

$$\eta_k = \frac{V_2 - V_1}{V_{str}} \quad (7)$$

And the input work given by:

$$W = m \frac{N}{60} (h_c - h_b) \quad (8)$$

B. Modeling of Condenser

Generally, the study of the heat transfer in a heat exchanger can be very difficult and can be simplified as following [10]:

i. Water Side Equations

Consider a Δz as single element, now it can be wrote the following energy balance given by [9]:

$$\frac{\partial}{\partial t}\left(\frac{\pi d^2}{4} \Delta z \rho_w c_{pw} T_w\right) = \pi d \Delta z \alpha (T_r - T_w) - \frac{\pi d^2}{4} V_w \rho_w c_{pw} (T_w|_{z+\Delta z} - T_w|_z) \quad (9)$$

Dividing eq. 9 by Δz and setting Δz →0, it can be obtained the governing equation of water side:

$$\frac{\pi d^2}{4}\rho_w c_{pw}\frac{\partial T_w}{\partial t} = \pi D\alpha(T_r - T_w) - \frac{\pi d^2}{4}V_w\rho_w c_{pw}\frac{\partial T_w}{\partial z} \quad (10)$$

ii. Refrigerant Side Equations

The equation of mass conservation and energy balance can be written as [10]:

$$\frac{d}{dt}(A\Delta z\rho) = \dot{m}|_z - \dot{m}|_{z+\Delta z} \quad (11)$$

Where

$$A = \frac{\pi D^2}{4} - N_{tubes}\frac{\pi d^2}{4}$$

By dividing of Δz and setting Δz →0 the mass balance given by:

$$\frac{d\rho}{dt} = -\frac{1}{A}\frac{\partial \dot{m}}{\partial z} \quad (12)$$

From the energy balance for the water side as:

$$\frac{d}{dt}(A\Delta z\rho u) = Av\rho(h_z - h_{z+\Delta z}) - \dot{Q}_{cond} \quad (13)$$

According to the Newton's law, the heat transferred between refrigerant and water given by:

$$\dot{Q}_{cond} = \pi d\Delta z\alpha(T_r - T_w) \quad (14)$$

Now, the heat exchanged amount between refrigerant and water is the same. By recall the definition of the specific internal energy that given by:

$$u = h - \frac{P}{\rho}$$

The energy balance becomes:

$$A\Delta zu\frac{d\rho}{dt} + A\Delta z\rho\frac{du}{dt} = Av\rho(h_z - h_{z+\Delta z}) - \pi d\Delta z\alpha(T_r - T_w) \quad (15)$$

By combining with the mass balance to get:

$$\frac{dh}{dt} = \frac{h}{\rho A}\frac{\partial \dot{m}}{\partial z} - v\frac{dh}{dz} - \frac{\pi d\alpha}{\rho A}(T_r - T_w) + \frac{d(P/\rho)}{dt} - \frac{d(P/\rho)}{\rho A}\frac{\partial \dot{m}}{\partial z} \quad (16)$$

iii. Modeling of Evaporator

In this study, the evaporator of type shell and tube flooded has been employed. The modeling of evaporator is rather simple. By applying the mass conservation and energy balance to the side of refrigerant as [11]:

$$\frac{d(\rho V_e)}{dt} = \dot{m}_{il} - \dot{m}_{ov} \tag{17}$$

$$\frac{d(\rho V_e u)}{dt} = \dot{m}_{il} h_a - \dot{m}_{ov} h_b + \dot{Q}_e \tag{18}$$

Rearranging eq. 17 and 18, the energy balance becomes [11]:

$$\rho V_e \frac{du}{dt} + u\frac{d(\rho V_e)}{dt} = \dot{m}_{il} h_a - \dot{m}_{ov} h_b + \dot{Q}_e \tag{19}$$

So

$$\rho V_e \frac{dh_e}{dt} + \dot{m}_{il}(h_e - h_a) + \dot{m}_{ov}(h_b - h_e) - \dot{Q}_e = \frac{P}{\rho}(\dot{m}_{il} - \dot{m}_{ov}) + V_e \frac{dP}{dt} \tag{20}$$

As mention above in the modeling the condenser, order to obtain the two evaporator governing equations, the pressure and enthalpy terms can be divided as following:

$$\rho V_e \frac{dh_e}{dt} + \dot{m}_{il}(h_e - h_a) + \dot{m}_{ov}(h_b - h_e) - \dot{Q}_e = 0 \tag{21}$$

And

$$V_e \frac{dP}{dt} + \frac{P}{\rho}(\dot{m}_{il} - \dot{m}_{ov}) = 0$$

iv. Modeling the Expansion Valve

Now, it can be supposed that the inertia effects neglected due to the valve acts as a static element with. Thus, the mass flow rate passing through the expansion device, the steady-state equation given by [11]:

$$\dot{m} = \rho C_v S_v \sqrt{\frac{2(P_{cond} - P_{eva})}{\rho}} \tag{22}$$

Also, the energy balance equation given by:

$$h_d = h_a \tag{23}$$

Theoretical Heat Pump Performance

It is conceivable to say that the heat pump must keep a heat output (QHP) at least equal to the required thermal demand, which is dependent on the calculated radiator water inlet/outlet temperature difference and mass flow rate, as follows:

$$Q_{HP} = m_{water} Cp_{water}(T_{in} - T_{out}) \tag{24}$$

For steady-state calculations, and considering a single-inlet, single-outlet condenser, the heat output (Q_{HP}). In this case, m_{R600a} is the mass flow rate of the working fluid; H3 and H4 are respectively, the fluid´s initial and final enthalpy [J/Kg], during the condenser phase.

$$Q_{HP} = -Q_{34} = m_{R600a}(H_3 - H_4) \quad (25)$$

Mass flow rate can be solved for the working fluid's mass flow rate:

$$m_{R600a} = \frac{Q_{HP}}{(H_3 - H_4)} \quad (26)$$

The data of $R600a$ obtained from [12] and inputted to MATLAB as functions, by using polinomial or linear regression techniques. Also the compressor work given by:

$$W_{th} = m_{R600a}(H_3 - H_2) \quad (27)$$

The theoretical COP_{th} given by [11]:

$$COP_{th} = \frac{Q_{HP}}{W_{Comp}} \quad (28)$$

With the compressor work and COP, all other relevant results can be obtained, as for instance the amount of energy removed/injected as (Q_{elect}) or the electricity consumption given by [10]:

$$Q_{elect} = Q_{HP} - W_{Comp} \quad (29)$$

4. RESULTS AND DISCUSSION

Experimental Results

Coefficient of Performance (COP)

Fig.(4) shows values for COP that were calculated from equation (1) for cases mentioned in Table(1), which shows that the values of COP were improved when by addition of Al_2O_3 to mineral oil. The actual COP was determined using the values of the power input and the cooling load. For comparison, the theoretical values are plotted in the figure. The theoretical COP is greater than the actual COP for all cases. Also Fig.(4) shows that the values of COP increases when the system was charging by the compositions of S_2, S_3 and S_4 about 19.1%, 11.6% and 7.3% respectively in comparison with of S_1. This result may be explained that when adding small of nanoparticles size of Al_2O_3 to the mineral oil it will reduces the power consumption by the compressor and a subcooling of the nanorefrigerant in the condenser will be noticed which in turn leads to increasing of the COP. Moreover, the obtained results showed a good agreement with the study of Reji, et al.[4]. The results showed that the refrigeration system COP increased by 19.6 % when used nanorefrigerant of R600a with Al_2O_3 nanoparticles instead of the conventional POE oil

Effect of Nanoparticles Size on Energy Consumption by the Compressor

Fig.(5) presented a comparison of the power consumption of the compressor between four cases under study, which shows a considerable reduction in power consumption when

minimizes the size of the nanolubricants. The power consumption reduction was about 21.8% when using S_2 but the reduction was about 12.72% and 7.27% was observed when using S_3 and S_4 respectively instead of S_1. Reji Kumar [4] concluded that when the nanolubricant is used instead of conventional POE oil, the compressor power consumption reduces about 11.5% which is in a good agreement with the obtained results.

Effect of Nanoparticles Size on Heat–Rejection Ratio (H.R.R)

The heat rejection-ratio (H.R.R) is a term often used to relate the rate of heat flow at the condenser to that of the evaporator. Fig.(6) shows that (H.R.R) decreases when used Nano-Refrigerant in the cycle. Moreover, the figure shows that using the mixtures of S_2, S_3 and S_4 the value (H.R.R) decrease about 2.73%, 1.45% and 0.85 % respectively in comparison with of S_1. Decreasing the values of (H.R.R) means that the cooling effect will increase and the work of compressor will decrease.

Refrigerating Effect

Fig. (7) shows the variations of refrigerating effect with the time. It was observed that the S_2 mixture has a higher refrigerating effect than the other cases when using nanoparticles in the refrigerant. Also, it is clearly shown from the figure that the refrigerating effect increases with an decrease nanoparticles size of Al_2O_3 that is due to occurrence of subcool in the condenser. As the heat transfer rate has been enhanced and the mass flow rate of nano-rerigerant has been increased, the work of compressor decreases and the power consumption reduces which in turn enhances the heat pump performance.

Effect of Nanoparticles Size on Work of Compressor

The variation of compressor work with compressor discharge temperature is shown in Fig.(8). The figure shows that the work of compression decreases as compressor discharge temperature increases. Also, it is shown from the figure that the mixture of S_2 has a lower work of compressor than the other cases. This is due to the fact that when the temperature of compressor discharge increases, the suction temperature of compressor also increases due to occurrence of subheat in suction line which causes increase in the mass of refrigerant circulated through the compressor per unit time hence decreases the work of compressor subsequently reduction in power consumption and performance enhancement heat pump.

Effect of Nanoparticles Size on the Condenser Outlet Temperatures

Fig. (9) represent the relationship between the evaporator temperature and outlet condenser temperature which shows that the evaporator temperature and outlet condenser temperature decrease when used the nano-refrigerant in the cycle. Also, it is observed that the decreasing of evaporator temperature and outlet condenser temperature is more when used the S_2 mixture than the other mixture cases. However, the addition of small size nanoparticles in refrigerant which increase the condenser heat transfer rate thus enhances performance improvement of heat pump.

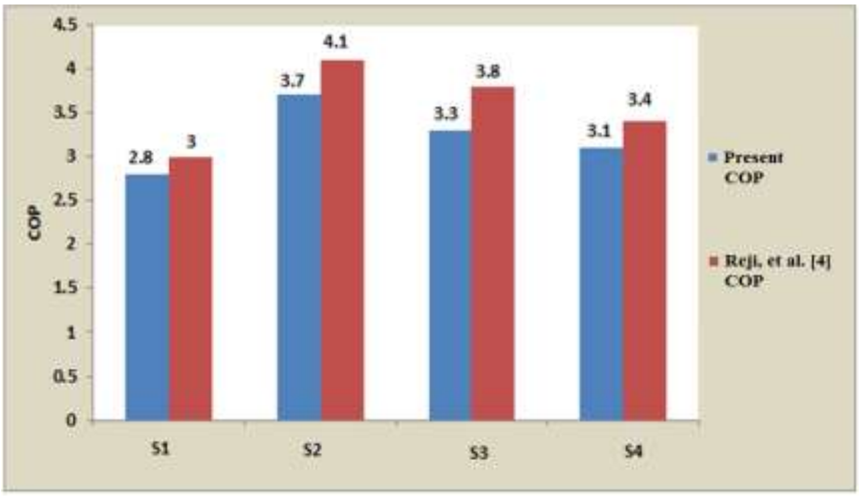


Figure 4 Experimental comparison of COP between the actual and theoretical values of the Reji Kumar [4] the four cases.

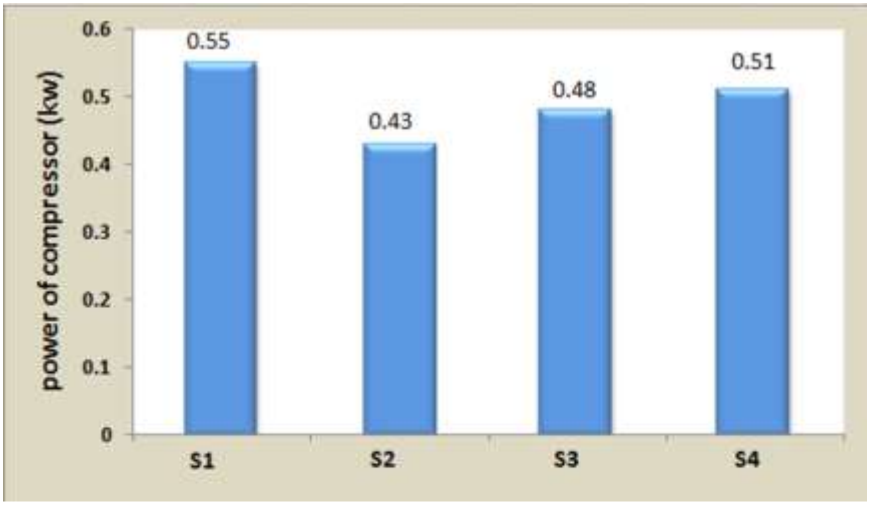


Figure 5 Experimental comparison of compressor power consumption between the four cases under study.

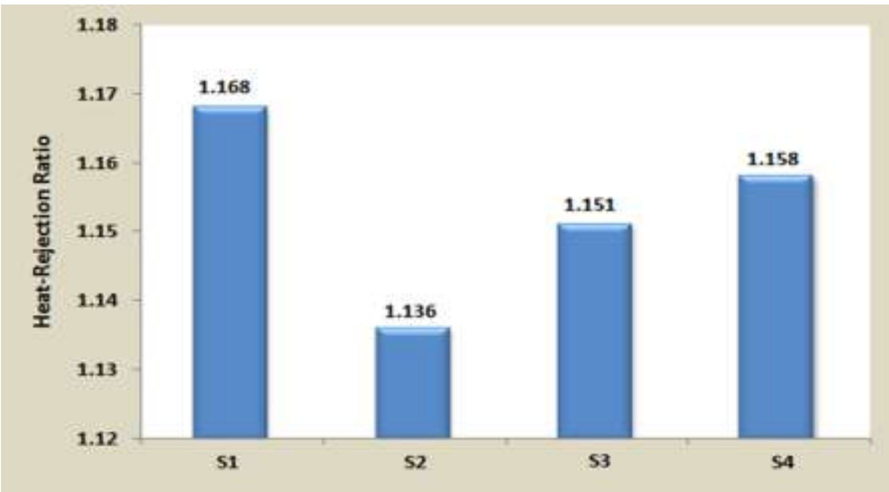


Figure 6 Experimental comparison of H.R.R between the four cases under study.

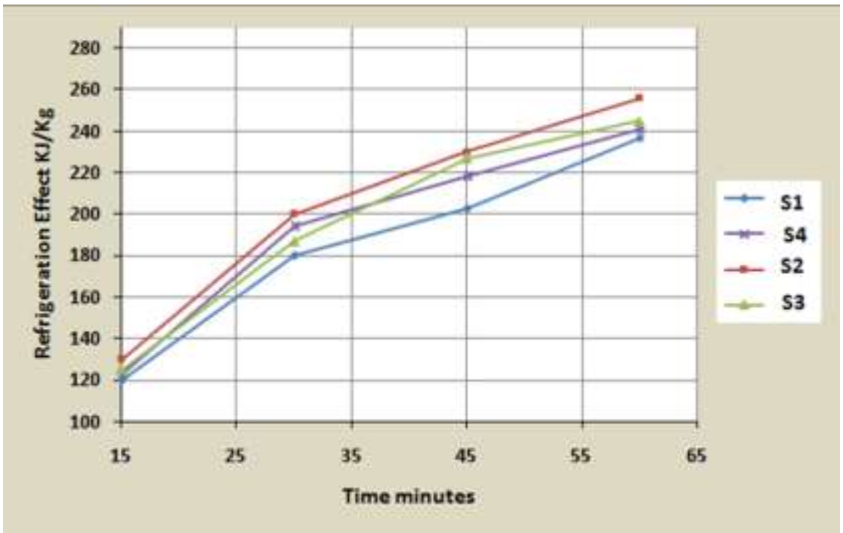


Figure 7 Experimental study the effect of nanoparticles size on refrigerating effect.

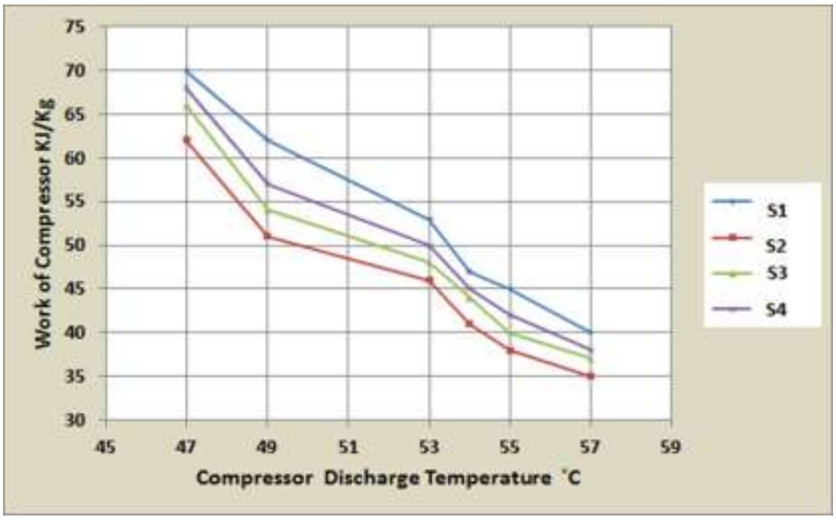


Figure 8 Experimental the effect of nanoparticles size on work of compressor.

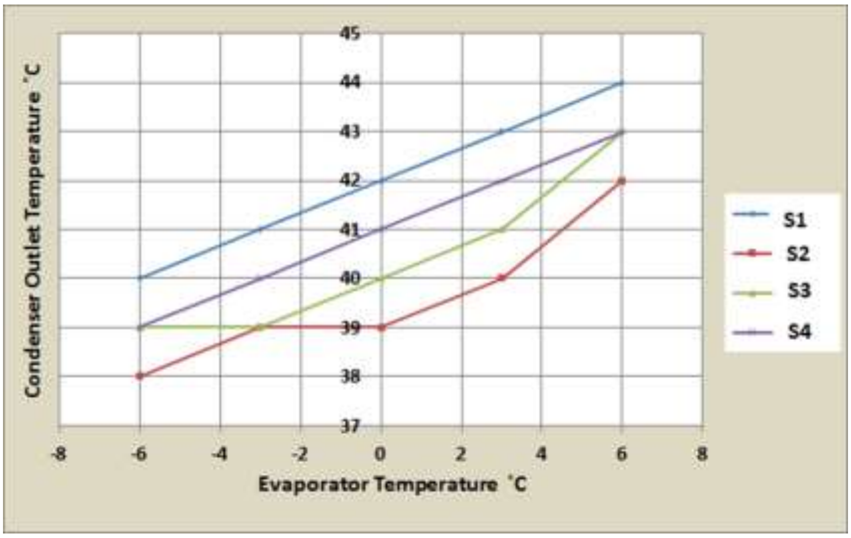


Figure 9 Experimental condenser outlet temperature Vs. evaporator temperature.

Theoretical Results

The aim of this study is the enhancement the thermophysical properties of $R600a$ by adding small size nanoparticles as a nanorefrigerant in the system of heating mode of heat pump model performance at various operating parameters. In the theoretical part of this study, the computer modelling of the heat pump components by using MATLAB program. The heat pump parameter performance incluids COP, HHR, compressor work compressor work and electricity consumption was studied. The following figures presents the effect of increasing the evaporator temperatures from -8 to 8 ^{o}C on the heat pump performance. Fig 10 shows that the heat pump parameter COP increased dramaticlly with evaporator temperatures, also it increasing by using the adding small of nanoparticles to the employed refrigerant. The maximum values of the COP found in using the type S4 of Mineral oil +50 nm nanoparticles size of Al_2O_3 which given increasing by (11%) greater than that the case of using S1 nanorefrigerants. The effect of using four types of nanorefrigerants on heat pump HHR plotted in Fig. (11), the results shown that the heat pump HHR decreasing with using the nanorefrigerants by (-12%) lower greater than that the case of using S1 nanorefrigerants at temperature 2 ^{o}C.

Fig.12 illustrated the effect of using four types of nanorefrigerants on the compressor work by therotical results, the results shows that the compressor work decreasing with using the nanorefrigerants instead of using the convential refrigerants with increasing the evaporator temperature. Also, Fig.13 presents the effect of evaporator temperature with using four types of nanorefrigerants on the electricity consumption, the results show that the electricity consumption decreasing with using the nanorefrigerants instead of using the convential refrigerants. For example, the electricity consumption decreasing by (-8%) lower greater than that the case of using S1 nanorefrigerants at evaporator temperature 2 ^{o}C. Figs. 14 present the effect of increasing the condenser temperature on the COP. Its indicated that the COP decreasing proportionality with the condenser temperature due to increasing the enthalpy of the compressor outlet, and this leads to increasing the compressor work. The effect of increasing the condenser temperature on the HHR was shown in Figs. 15. The results indicated that the HHR increases with increasing the condenser temperature due to increasing the enthalpy of the condenser inlet, and this leads to increasing the rate of heat rejection at the condenser. The effect of increasing the condenser temperature on the HHR was shown in Figs. 16. The results indicated that the HHR increases with increasing the condenser temperature due to increasing the enthalpy of the condenser inlet, and this leads to increasing the rate of heat rejection at the condenser. Finally, Fig.17 plotted the relationship between the electricity consumption with condenser temperature for the four cases of using nanorefrigerants were employed in this study. The results showed that the increasing the condenser temperature from 35 to 55 ^{o}C, the electricity consumption increasing by (75 %) in case of using S4 and by (70 %) in case of using S1 nanorefrigerants. The theoretical values compared with the experimental data as shown in Fig. 18. The figure showed that the values of COP were improved when by addition of Al_2O_3 to mineral oil. Moreover, the obtained results showed a good agreement with the data that was given by the theoretical work.

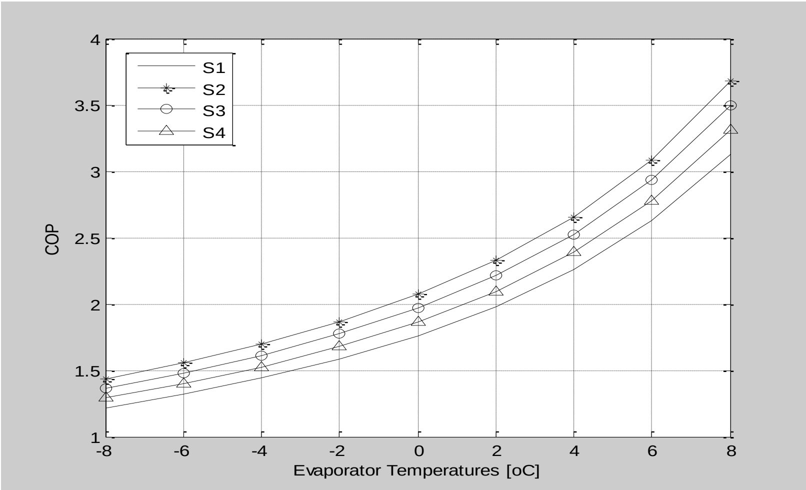


Figure 10 Theoretical variation COP with evaporator temperature.

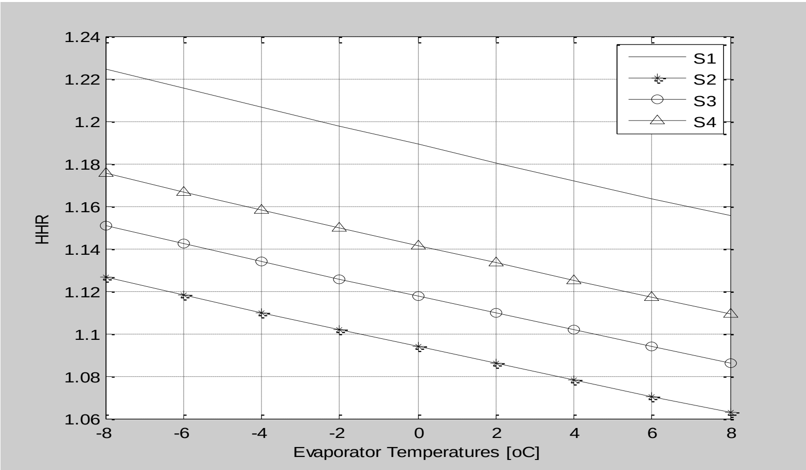


Figure 11 Theoretical variation HHR with evaporator temperature.

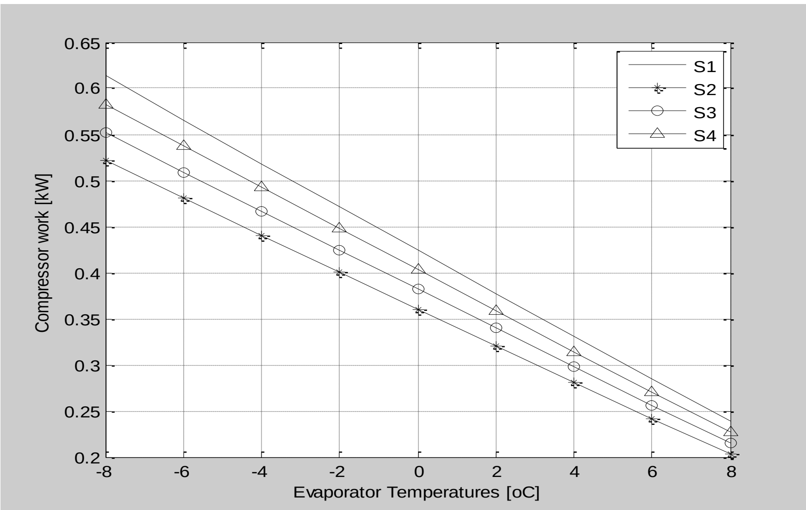


Figure 12 Theoretical variation compressor work with evaporator temperature.

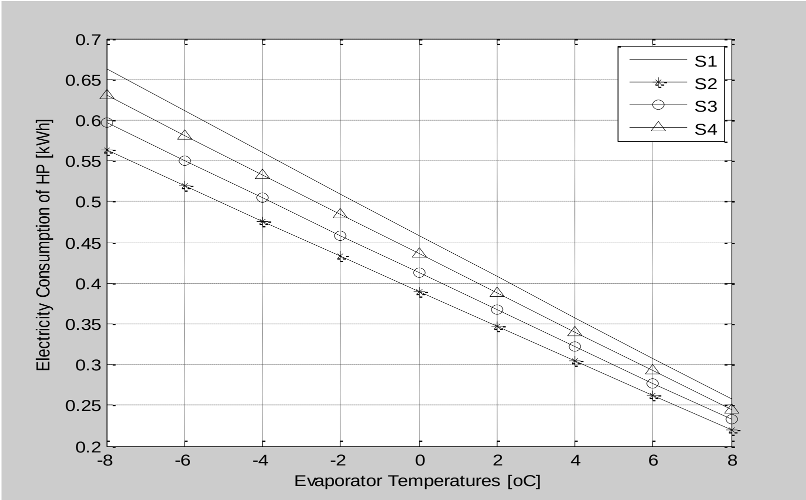


Figure 13 Theoretical variation electricity consumption with evaporator temperature.

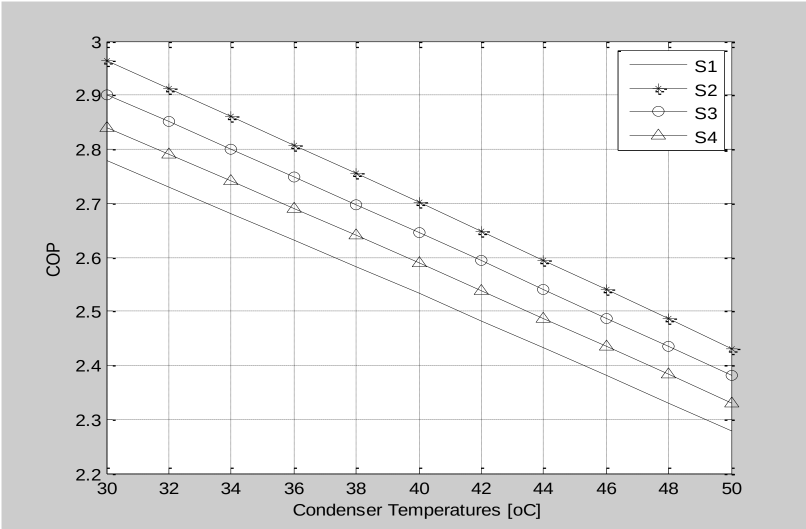


Figure 14 Theoretical variation COP with condenser temperature.

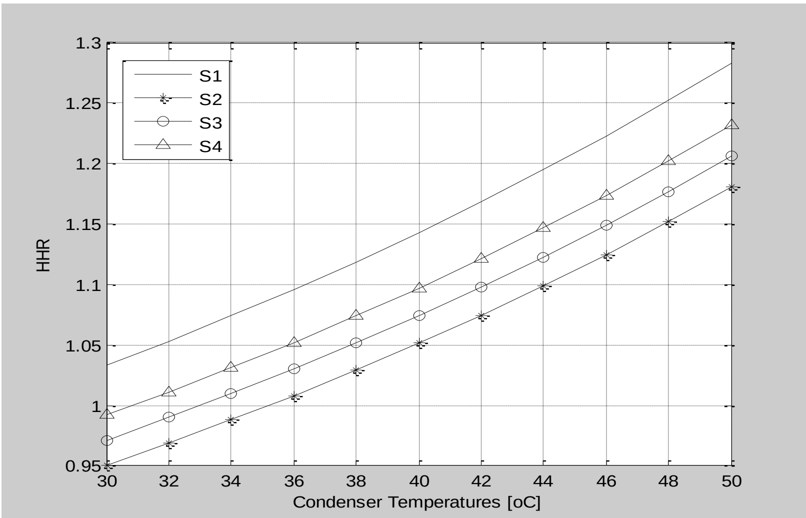


Figure 15 Theoretical variation HHR with condenser temperature.

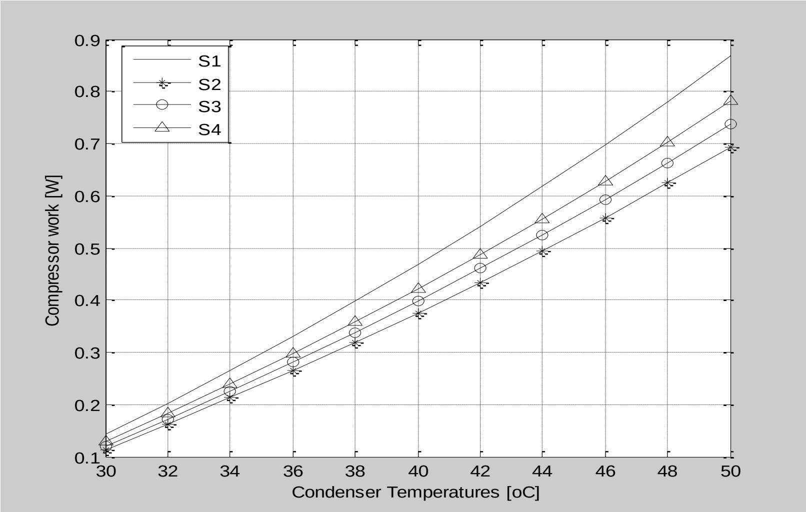


Figure 16 Theoretical variation compressor work with condenser temperature.

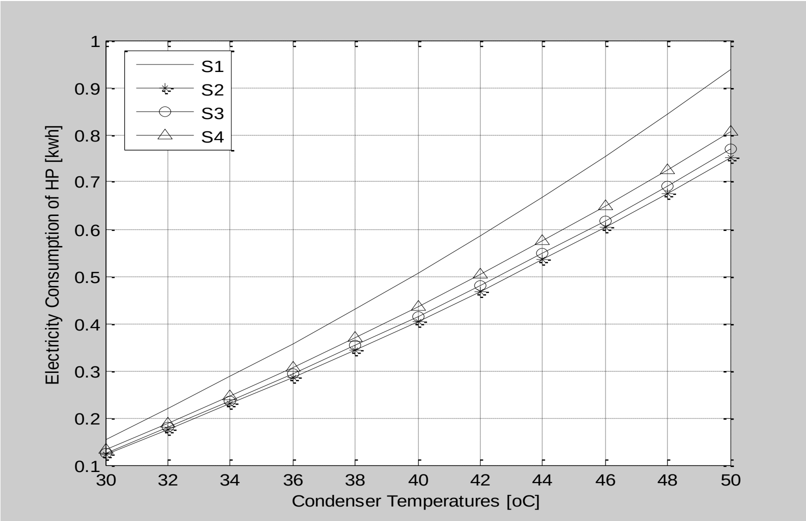


Figure 17 Theoretical variation electricity consumption with condenser temperature.

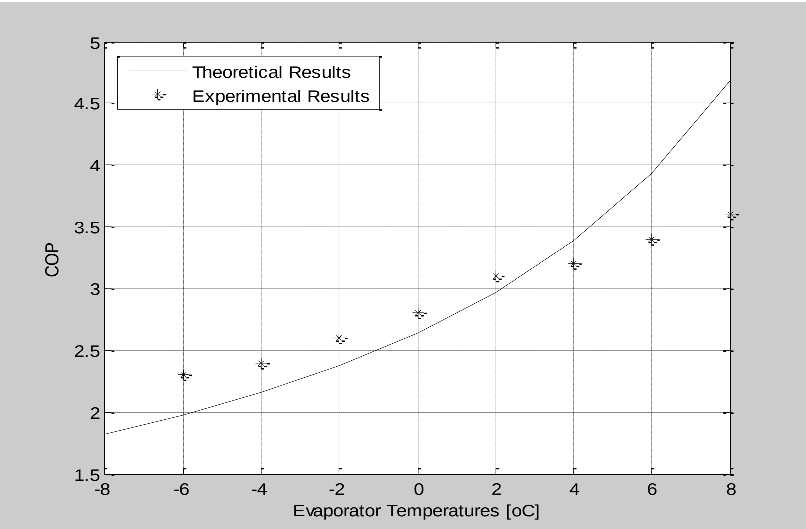


Figure 18 Presents theoretical and experimental comparison results of the COP.

5. CONCLUSIONS

The present study has concentrated on the effect of nanoparticles size of Al_2O_3 on the performance of the heat pump for giving an indication for improvements the performance of the heat pump. The main conclusions from this work were that the refrigerating effect increases with the decrease in nanoparticles size of Al_2O_3. Among Four cases which have been considered for this experimental study, it was found that the using of mineral oil with 20 nm nanoparticles of Al_2O_3 leads to COP improvement by 19.1%, and reduces the compressor power consumption by about 21.8%, also, reduces the heat rejection by about 2.73% in comparison with the case of hermetic compressor was filled with SUNISO 3GS oil (mineral oil). Moreover, the condenser outlet and evaporator temperatures will be more decreased

NOMENCLATURES

A	refrigerant flow sectional area (m^2)
m	flow rate of mass (kg/s)
V	flow rate of volume (m^3/s)
Q	heat transfer rate (W)
W	work of compressor (W)
Z	condenser axial coordinate, (m)
H	enthalpy of refrigerant (J/kg)
U	internal energy of refrigerant (J/kg)
P	pressure of refrigerant (Pa)
V	specific volume of refrigerant (m3/kg)
T	temperature (K)
C_p	specific heat at constant pressure (J/kgK)
C_v	specific heat at constant volume (J/kgK)
K	thermal conductivity (W/mK)
h_d, h_a	enthalpy at the expansion valve inlet and outlet.
h_1	inlet refrigerant enthalpy of compressor(kJ/kg)
h_2	outlet refrigerant enthalpy of compressor (kJ/kg)
h_3	inlet refrigerant enthalpy of evaporator (kJ/kg)
h_4	outlet refrigerant enthalpy of evaporator (kJ/kg)
$\dot{m}_{R500a}$	refrigerant flow rate (kg/s)
h_c	refrigerant enthalpy at the compressor outlet (kJ/kg)
m_{i1}	mass flow rate of evaporator inlet, (kg/s)
m_{ov}	mass flow rate of evaporator outlet, (kg/s)
C_v	orifice flow coefficient
S_v	cross sectional area of the orifice hole.

Greek Letters

μ	fluid dynamic viscosity, Pa.sec.
ν	fluid kinematic viscosity, m^2/sec.
ρ	density of the air, kg/m^3.

Subscripts

a inlet of evaporator
b inlet of compressor
c inlet of condenser
d inlet of expansion valve
$c_{,is}$ outlet of isentropic compression
e evaporator
il liquid inlet
ov vapour outlet
str stroke
suc suction
sv valve suction
th theoretical
v valve
w water

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